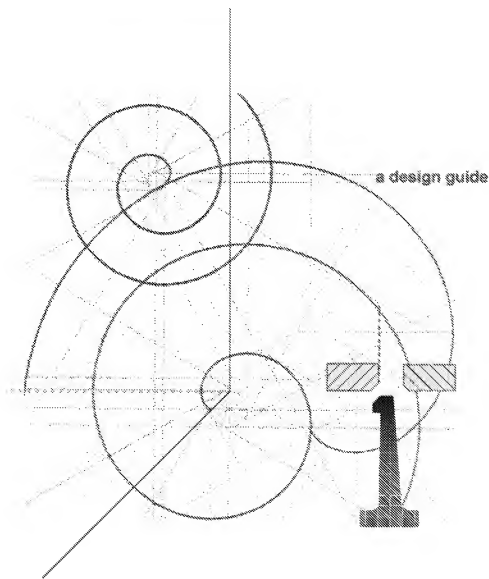
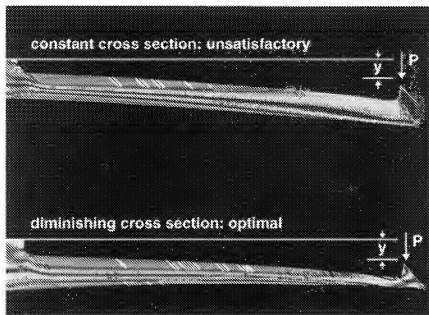




Bayer MaterialScience

# Snap-Fit Joints for Plastics





The illustration above shows a photograph of two snap-fit models taken in polarized light; both have the same displacement ( $y$ ) and deflective force ( $P$ ).

**Top:** The cantilever arm of unsatisfactory design has a constant cross section. The non-uniform distribution of lines (fringes) indicates a very uneven strain in the outer fibers. This design uses 17% more material and exhibits 46% higher strain than the optimal design.

**Bottom:** The thickness of the optimal snap-fit arm decreases linearly to 30% of the original cross-sectional area. The strain in the outer fibers is uniform throughout the length of the cantilever.

## A Snap Joints/General

- Common features
- Types of snap joints
- Comments on dimensioning

## B Cantilever Snap Joints

- Hints for design Calculations
- Permissible undercut
- Deflection force, mating force
- Calculation examples

## C Torsion Snap Joints

- Deflection
- Deflection force

## D Annular Snap Joints

- Permissible undercut
- Mating force
- Calculation example

## E Both Mating Parts Elastic

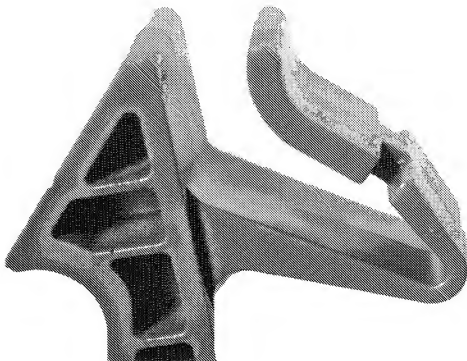
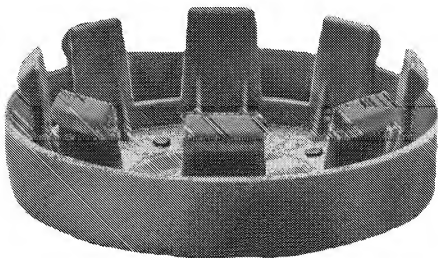
## F Symbols

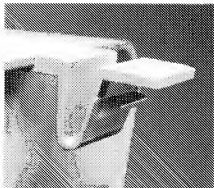
## Common features

Snap joints are a very simple, economical and rapid way of joining two different components. All types of snap joints have in common the principle that a protruding part of one component, e.g., a hook, stud or bead is deflected briefly during the joining operation and catches in a depression (undercut) in the mating component.

After the joining operation, the snap-fit features should return to a stress-free condition. The joint may be separable or inseparable depending on the shape of the undercut; the force required to separate the components varies greatly according to the design. It is particularly important to bear the following factors in mind when designing snap joints:

- Mechanical load during the assembly operation.
- Force required for assembly.





## Types of snap joints

A wide range of design possibilities exists for snap joints.

In view of their high level of flexibility, plastics are usually very suitable materials for this joining technique.

In the following, the many design possibilities have been reduced to a few basic shapes. Calculation principles have been derived for these basic designs.

**The most important are:**

- **Cantilever snap joints**

The load here is mainly flexural.

- **U-shaped snap joints**

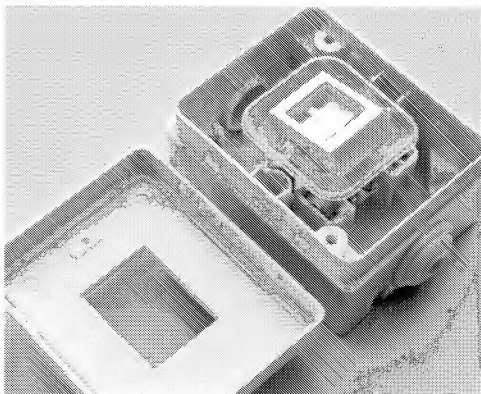
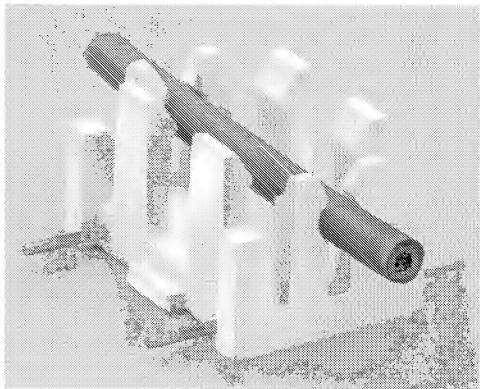
A variation of the cantilever type.

- **Torsion snap joints**

Shear stresses carry the load.

- **Annular snap joints**

These are rotationally symmetrical and involve multiaxial stresses.



## Cantilever snap joints

The four cantilevers on the control panel module shown in Fig. 1 hold the module firmly in place in the grid with their hooks, and yet it can still be removed when required. An economical and reliable snap joint can also be achieved by rigid lugs on one side in combination with snap-fitting hooks on the other (Fig. 2). This design is particularly effective for joining two similar halves of a housing which need to be easily separated. The positive snap joint illustrated in Fig. 3 can transmit considerable forces. The cover can still be removed easily from the chassis, however, since the snap-fitting arms can be re-leased by pressing on the two tongues in the direction of the arrow.

The example shown in Fig. 4 has certain similarities with an annular snap joint. The presence of slits, however, means that the load is predominantly flexural; this type of joint is therefore classified as a "cantilever arm" for dimensioning purposes.

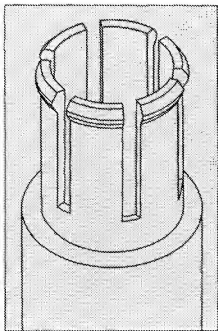


Fig 4: Discontinuous annular snap joint

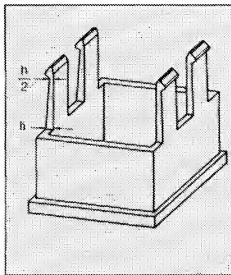


Fig 1: Module for control panels with four cantilever lugs

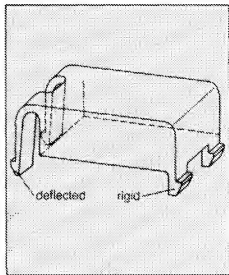


Fig 2: Cap with two cantilever and two rigid lugs

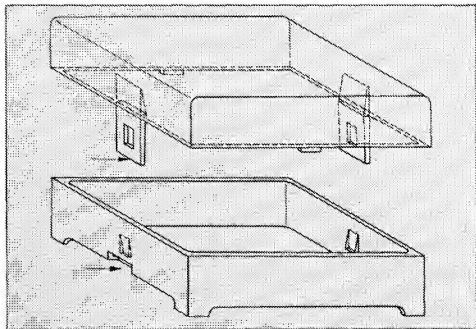


Fig 3: Separable snap joints for a chassis cover

## Torsion snap joints

The torsion snap joint of the design shown for an instrument housing in Fig. 5 is still uncommon in thermoplastics, despite the fact that it, too, amounts to a sophisticated and economical joining method. The design of a rocker arm whose deflection force is given largely by torsion of its shaft permits easy opening of the cover under a force  $P$ ; the torsion bar and snap-fitting rocker arm are integrally molded with the lower part of the housing in a single shot.

## Annular snap joints

A typical application for annular snap joints is in lamp housings (Fig. 6). Here, quite small undercuts give joints of considerable strength.

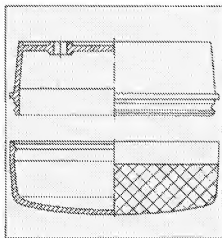


Fig 6: A continuous annular snap joint offers a semi-hermetic seal and is better for single assembly applications

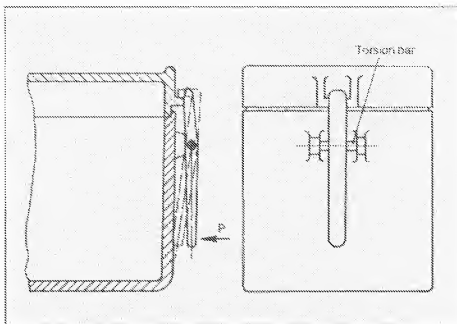
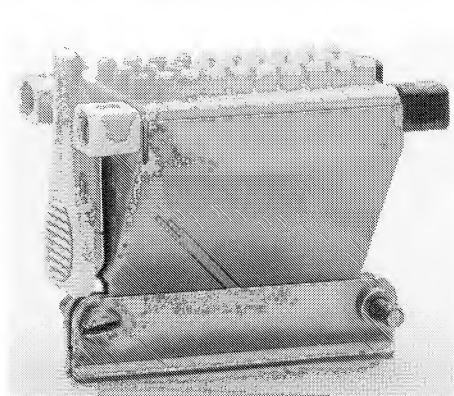


Fig 5: Torsion snap joint on a housing made of Makrolon polycarbonate



## Combination of different snap joint systems –

The traffic light illustrated in Fig. 7 is an example of an effective design for a functional unit. All the components of the housing are joined together by snap joint.

### Details:

- Housing and front access door engage at the fulcrum 1a. The lugs 1b (pressure point) hold the door open, which is useful for changing bulbs.
- The cantilever hook 2 locks the door. The door can be opened again by pressing the hook through the slit in the housing at 2.
- The reflector catches at three points on the periphery. Either a snap-fittin hook 3a or a pressure point 3b may be chosen here, so that there is polygonal deformation of the inner ring of the housing.
- The lens in the front door is either produced in the second of two moldings 4a or, if a glass lens is desired, this can be held by several cantilever snaps 4b.
- The sun visor engages at 5 like a bayonet catch. Good service-ability and low-cost production can be achieved with carefully thought-out designs such as this.

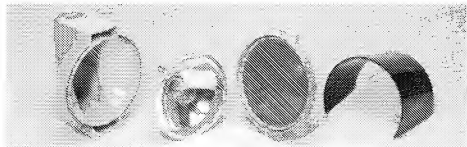
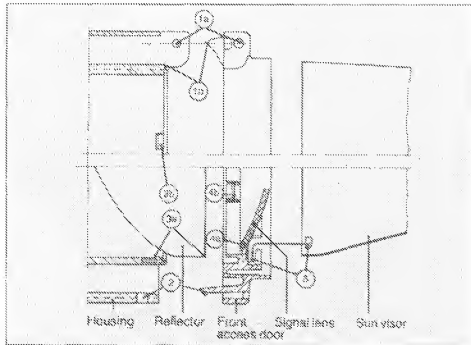


Fig. 7: Cross-sectional sketch (above) and photo (below) of a traffic light made of Makrolon®/polycarbonate. All the components are held together entirely by means of snap joints

## Assumptions

The calculation procedures applicable to various types of joints are briefly described on the following pages, but in such a way as to be as general as possible. The user can therefore apply this information to types of joints not dealt with directly.

In all the snap-fit designs that follow, it is assumed initially that one of the mating parts remains rigid. This assumption represents an additional precaution against material failure. If the two components are of approximately equal stiffness, half the deflection can be assigned to each part. If one component is more rigid than the other and the total strength available is to be utilized to the fullest, the more complex procedure described in Section E must be adopted.

What is said in the remainder of the brochure takes into account the fact that the plastics parts concerned are, for brief periods, subjected to very high mechanical loads. This means that the stress-strain behavior of the material is already outside the linear range and the ordinary modulus of elasticity must therefore be replaced by the strain dependent secant modulus.

## Design Hints

A large proportion of snap joints are basically simple cantilever snaps (Fig. 8), which may be of rectangular or of a geometrically more complex cross section (see Table 1).

It is suggested to design the finger so that either its thickness ( $h$ ) or width ( $b$ ) tapers from the root to the hook; in this way the load-bearing cross section at any point bears a more appropriate relation to the local load. The maximum strain on the material can therefore be reduced, and less material is needed.

Good results have been obtained by reducing the thickness ( $h$ ) of the cantilever linearly so that its value at the end of the hook is equal to one-half the value at the root; alternatively, the finger width may be reduced to one-quarter of the base value (see Table 1, designs 2 and 3). With the designs illustrated in Table 1, the vulnerable cross section is always at the root (see also Fig. 8, Detail A). Special attention must therefore be given to this area to avoid stress concentration.

Fig. 9 graphically represents the effect the root radius has on stress concentration. At first glance, it seems that an optimum reduction in stress concentration is obtained using the ratio  $R/h$  as 0.6 since only a marginal reduction occurs after this point. However, using  $R/h$  of 0.6 would result in a thick area at the intersection of the snap-fit arm and its base. Thick sections will usually result in sinks and/or voids which are signs of high residual stress. For this reason, the designer should reach a compromise between a large radius to reduce stress concentration and a small radius to reduce the potential for residual stresses due to the creation of a thick section adjacent to a thin section. Internal testing shows that the radius should not be less than 0.015 in. in any instance.

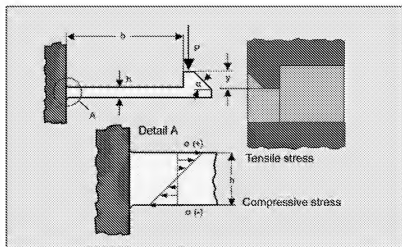


Fig. 8: Simple snap-fitting hook

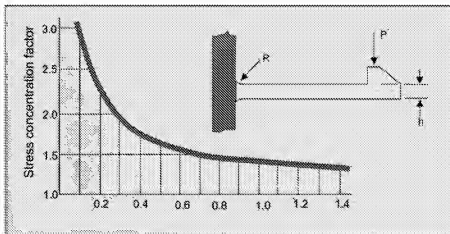
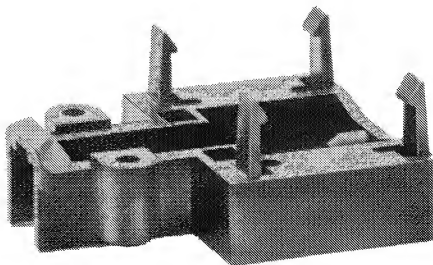


Fig. 9: Effects of a fillet radius on stress concentration



## Calculations

Shape of the cross section Type of design		A	B	C	D
(Permissible) deflection	1 	$y = 0.67 \cdot \frac{\epsilon \cdot l^2}{h}$	$y = \frac{a+b}{2a+b} \cdot \frac{\epsilon \cdot l^2}{h}$	$y = K_{12} \cdot \frac{\epsilon \cdot l^2}{r_1}$	$y = \frac{1}{3} \cdot \frac{\epsilon \cdot l^2}{c_{23}}$
	2 	$y = 1.09 \cdot \frac{\epsilon \cdot l^2}{h}$	$y = 1.64 \cdot \frac{a+b_{12}}{2a+b} \cdot \frac{\epsilon \cdot l^2}{h}$	$y = 1.64 \cdot K_{23} \cdot \frac{\epsilon \cdot l^2}{r_1}$	$y = 0.55 \cdot \frac{\epsilon \cdot l^2}{c_2}$
	3 	$y = 0.86 \cdot \frac{\epsilon \cdot l^2}{h}$	$y = 1.28 \cdot \frac{a+b_{31}}{2a+b} \cdot \frac{\epsilon \cdot l^2}{h}$	$y = 1.28 \cdot K_{32} \cdot \frac{\epsilon \cdot l^2}{r_1}$	$y = 0.43 \cdot \frac{\epsilon \cdot l^2}{c_{23}}$
Deflection force		$P = \frac{Z}{6} \cdot \frac{E \cdot \epsilon}{l}$	$P = \frac{Z}{12} \cdot \frac{h^3 \cdot a^3 + 4ab_{12} + b^3}{2a+b} \cdot \frac{E \cdot \epsilon}{l}$	$P = Z_{12} \cdot \frac{E \cdot \epsilon}{l}$	$P = Z_{32} \cdot \frac{E \cdot \epsilon}{l}$
Subscript numbers in parenthesis designate the note to refer to.					

Table 1: Equations for dimensioning cantilevers

## Symbols

- $y$  = (permissible) deflection (=undercut)  
 $E$  = (permissible) strain in the outer fiber at the root; in formulae:  $E$  as absolute value = percentage/100 (see Table 2)  
 $l$  = length of arm  
 $h$  = thickness at root  
 $b$  = width at root  
 $c$  = distance between outer fiber and neutral fiber (center of gravity)  
 $Z$  = section modulus  $Z = I/c$ , where  $I$  = axial moment of inertia  
 $E_s$  = secant modulus (see Fig. 16)  
 $P$  = (permissible) deflection force  
 $K$  = geometric factor (see Fig. 10)

## Notes

- These formulae apply when the tensile stress is in the small surface area  $b$ . If it occurs in the larger surface area  $a$ , however,  $a$  and  $b$  must be interchanged.
- If the tensile stress occurs in the convex surface, use  $K_2$  in Fig. 10; if it occurs in the concave surface, use  $K_1$ , accordingly.
- $c$  is the distance between the outer fiber and the center of gravity (neutral axis) in the surface subject to tensile stress.

- The section modulus should be determined for the surface subject to tensile stress. Section moduli for cross-section shape type C are given in Fig. 11. Section moduli for other basic geometrical shapes are to be found in mechanical

Permissible stresses are usually more affected by temperatures than the associated strains. One preferably determines the strain associated with the permissible stress at room temperature. As a first approximation, the computation may be based on this value regardless of the temperature. Although the equations in Table 1 may appear unfamiliar, they are simple manipulations of the conventional engineering equations to put the analysis in terms of permissible strain levels.

## Geometric factors K and Z for ring segment

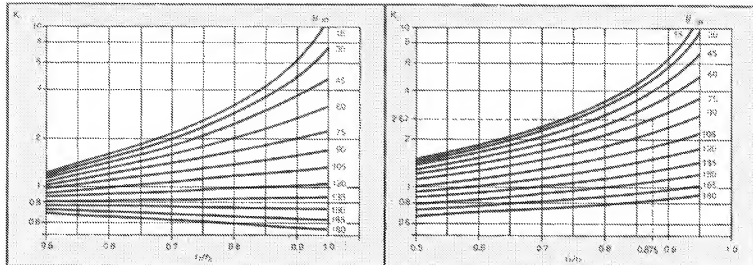


Fig 10: Diagrams for determining K1 and K2 for cross-sectional shape type C in Table 1.

K1: Concave side under tensile load, K2: Convex side under tensile load

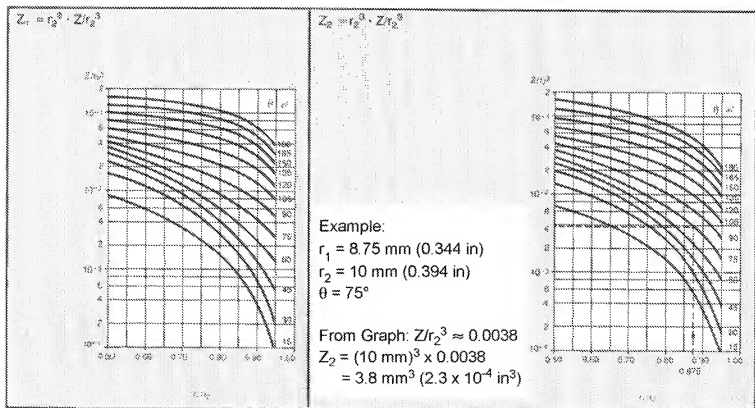


Fig 11: Graphs for determining the dimensionless quantity ( $Z/r_2^3$ ) used to derive the section modulus ( $Z$ ) for cross-sectional shape C in Table 1.

Z1: concave side under tensile stress, Z2: convex side under tensile stress

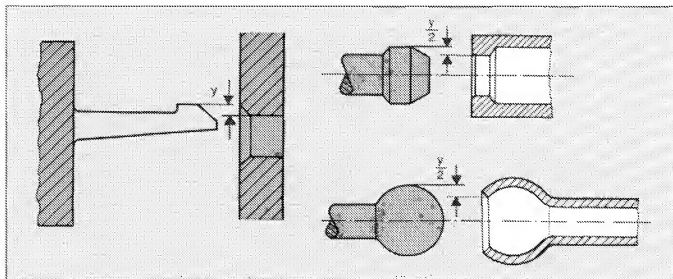


Fig 12: Undercut for snap joints

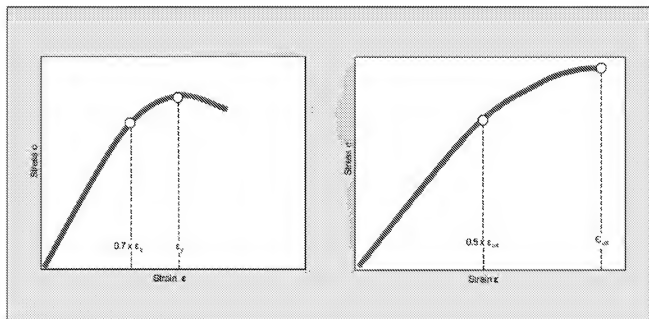


Fig 13: Determination of the permissible strain for the joining operation (left: material with distinct yield point; right: glass-fiber-reinforced material without yield point)

## Permissible undercut

The deflection  $y$  occurring during the joining operation is equal to the undercut (Fig. 12).

The permissible deflection  $y$  (permissible undercut) depends not only on the shape but also on the permissible strain  $E$  for the material used.

In general, during a single, brief snap-fitting operation, partially crystalline materials may be stressed almost to the yield point, amorphous ones up to about 70% of the yield strain.

Glass-fiber-reinforced molding compounds do not normally have a distinct yield point. The permissible strain for these materials in the case of snap joints is about half the elongation at break (see Fig. 13)

## Deflection force

Using the equations given in Table 1, the permissible deflection  $y$  can be determined easily even for cross sections of complex shapes. The procedure is explained with the aid of an example which follows.

A particularly favorable form of snap-fitting arm is design 2 in Table 1, with the thickness of the arm decreasing linearly to half its initial value. This version increases the permissible deflection by more than 60% compared to a snap-fitting arm of constant cross section (design 1).

Complex designs such as that shown in Fig. 15 may be used in applications to increase the effective length. Polymers Division Design Engineering Services would be pleased to assist you in a curved beam analysis if you choose this type of design.

The deflection force  $P$  required to bend the finger can be calculated by use of the equations in the bottom row of Table 1 for cross sections of various shapes.

$E_s$  is the strain dependent modulus of elasticity or "secant modulus" (see Fig. 14).

Values for the secant modulus for various Bayer engineering plastics can be determined from Fig. 16. The strain value used should always be the one on which the dimensioning of the undercut was based.

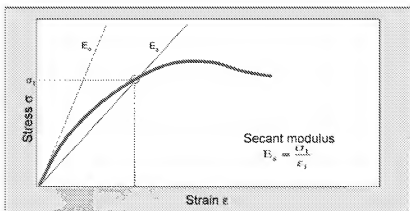


Fig. 14: Determination of the secant modulus

## Permissible short term strain limits at 23°C (73°F)

### Unreinforced

Apec®	High Heat PC	4%
Bayblend®	PC/ABS	2.5%
Makroblend®	Polycarbonate Blends	3.5%
Makrolon®	PC	4%

### Glass-Fiber-Reinforced (%Glass)

Makrolon®(10%)	PC	2.2%
Makrolon®(20%)	PC	2.0%

Table 2: General guide data for the allowable short-term strain for snap joints (single joint - ing operation); for frequent separation and rejoining, use about 60% of these values

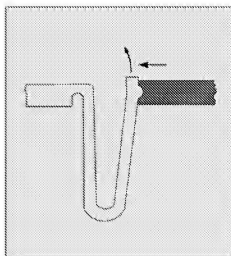


Fig. 15: U-shaped snap-fitting arm for a lid fastening

## Polyurethane Snap-Fits

Snap-fits are possible using certain polyurethane systems. For more information call Polymers Design Engineering at 412-777-4952.

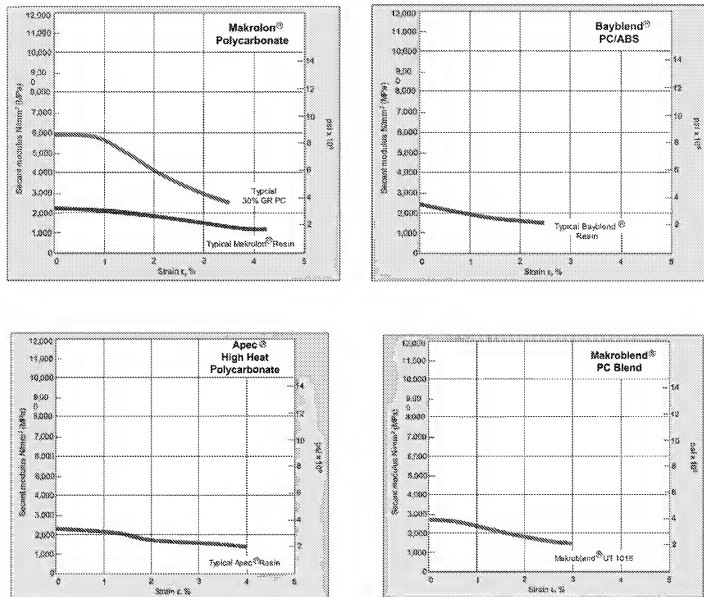


Fig. 16: Secant Modulus for Bayer engineering plastics at 23°C (73°F)

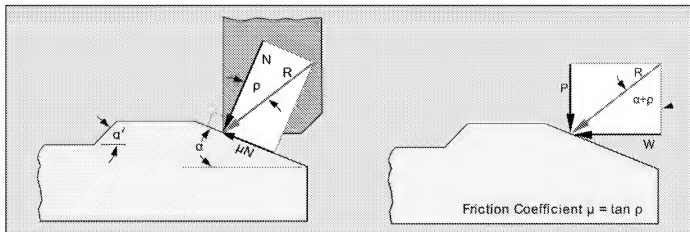


Fig. 17: Relationship between deflection force and mating force

## Mating Force

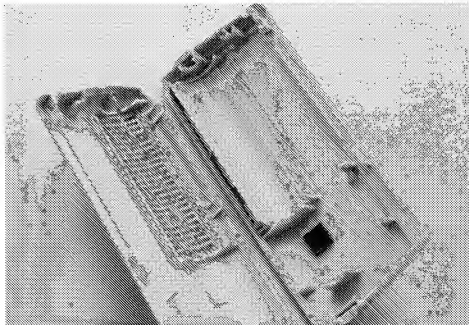
During the assembly operation, the deflection force  $P$  and friction force  $F$  have to be overcome (see Fig. 17).

The mating force is given by:

$$W = P \cdot \tan(\alpha + p) = P \cdot \frac{\mu + \tan \alpha}{1 - \mu \tan \alpha}$$

The value for  $1 - \mu \tan \alpha$  can be taken directly from Fig. 18. Friction coefficients for various materials are given in Table 3.

In case of separable joints, the **separation force** can be determined in the same way as the mating force by using the above equation. The angle of inclination to be used here is the angle  $\alpha'$



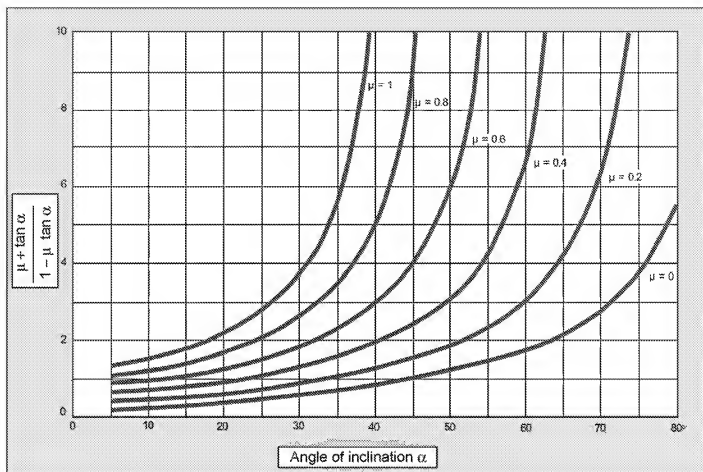


Figure 18: Diagram for determining  $\frac{\mu + \tan \alpha}{1 - \mu \tan \alpha}$

The figures depend on the relative speed of the mating parts, the pressure applied and on the surface quality. Friction between two different plastic materials gives values equal to or slightly below those shown in Table 3. With two components of the same plastic material, the friction coefficient is generally higher. Where the factor is known, it has been indicated in parentheses.

PTFE	0.12-0.22
PE rigid	0.20-0.25 (x 2.0)
PP	0.25-0.30 (x 1.5)
POM	0.20-0.35 (x 1.5)
PA	0.30-0.40 (x 1.5)
PBT	0.35-0.40
PS	0.40-0.50 (x 1.2)
SAN	0.45-0.55
PC	0.45-0.55 (x 1.2)
PMMA	0.50-0.60 (x 1.2)
ABS	0.50-0.65 (x 1.2)
PE flexible	0.55-0.60 (x 1.2)
PVC	0.55-0.60 (x 1.0)

**Table 3: Friction coefficient,  $\mu$ .**  
(Guide data from literature for the coefficients of friction of plastics on steel.)

## Calculation example I snap-fitting hook

This calculation is for a snap-fitting hook of rectangular cross section with a constant decrease in thickness from  $h$  at the root to  $h/2$  at the end of the hook (see Fig. 19). This is an example of design type 2 in Table 1 and should be used whenever possible to permit greater deformation and to save material.

### Given:

- Material = Makrolon® polycarbonate
- Length ( $l$ ) = 19 mm (0.75 in)
- Width ( $b$ ) = 9.5 mm (0.37 in)
- Undercut ( $y$ ) = 2.4 mm (0.094 in)
- Angle of inclination ( $\alpha$ ) =  $30^\circ$

### Find:

- Thickness  $h$  at which full deflection  $y$  will cause a strain of one-half the permissible strain.
- Deflection force  $P$
- Mating force  $W$

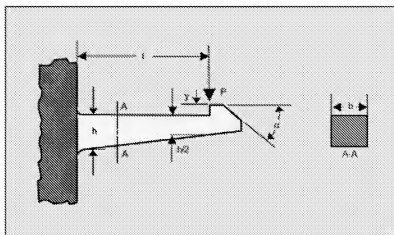


Fig. 19: Snap-fitting hook, design type 2, shape A

### Solution:

- Determination of wall thickness  $h$

Permissible strain from Table 2:  $\epsilon_m = 4\%$

Strain required here  $\epsilon = 1/2 \epsilon_m = 2\%$

Deflection equation from Table 1, type 2, shape A:

Transposing in terms of thickness

$$y = 1.09 \frac{\epsilon l^2}{h}$$

$$h = \frac{1.09 \epsilon l^2}{y}$$

$$= \frac{1.09 \times 0.02 \times 19^2}{2.4}$$

$$= 3.28 \text{ mm (0.13 in)}$$

### b. Determination of deflection force $P$

$$\text{Deflection force equation from Table 1, cross section A: } P = \frac{bh^3}{6} \cdot \frac{E_s \epsilon}{l}$$

From Fig. 16 at  $\epsilon = 2.0\%$

$E_s = 1,815 \text{ N/mm}^2$  (264,000 psi)

$$P = \frac{9.5 \text{ mm} \times (3.28 \text{ mm})^3}{6} \cdot \frac{1,815 \text{ N/mm}^2 \times 0.02}{19 \text{ mm}}$$

$$= 32.5 \text{ N (7.3 lb)}$$

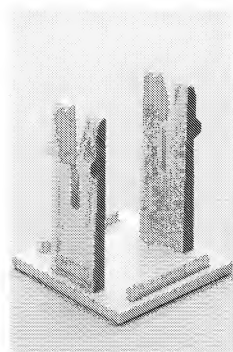
### c. Determination of mating force $W$

$$W = P \cdot \frac{\mu + \tan \alpha}{1 - \mu \tan \alpha}$$

Friction coefficient from Table 3 (PC against PC)  $\mu = 0.50 \times 1.2 = 0.6$

$$\text{From Fig. 18: } \frac{\mu + \tan \alpha}{1 - \mu \tan \alpha} = 1.8 \text{ For } \mu = 0.6 \text{ and } \alpha = 30^\circ$$

$$W = 32.5 \text{ N} \times 1.8 = 58.5 \text{ N (13.2 lb)}$$



## Calculation example II snap-fitting hook

This calculation example is for a snap-fitting hook with a segmented ring cross section decreasing in thickness from  $h$  at the root to  $h/2$  at the end of the hook (see Fig. 20). This is design type 2, shape C in Table 1.

This taper ratio should be used when possible to evenly distribute stresses during arm deflection. It also reduces material usage.

### Given:

- Material = Bayblend® PC/ABS
- Length ( $l$ ) = 25.4 mm (1.0 in)
- Angle of arc ( $\theta$ ) =  $75^\circ$
- Outer radius ( $r_2$ ) = 20 mm (0.787 in)
- Inner radius ( $r_1$ ) = 17.5 mm (0.689 in)
- Thickness ( $h$ ) = 2.5 mm (0.1 in)
- $\phi = \theta/2 = 37.5^\circ$

### Find:

- The maximum allowable deflection for a snap-fit design which will be assembled and unassembled frequently.

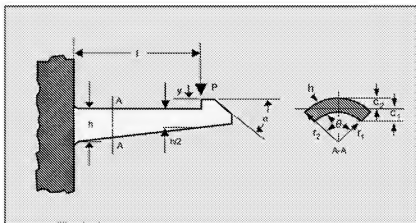


Fig. 20: Snap-fitting hook, design type 2, shape C

### Solution:

The permissible strain for a one-time snap-fit assembly in Bayblend® resin is 2.5%. Since the design is for frequent separation and rejoining, 60% of this value should be used or  $\epsilon_{\text{perm}} = (0.6)(2.5\%) = 1.5\%$ .

Deflection equation from Table 1, type 2, shape C:  $y = 1.64 K_{(2)} \frac{\epsilon l^2}{r_2}$

The variable for  $K_{(2)}$  can be obtained from the curves in Fig. 10. Note that if the member is deflected so that the tensile stress occurs in the convex surface, the curve for  $K_1$  should be used; if it occurs in the concave surface,  $K_2$  should be used. In this case, the tensile stress will occur in the convex surface, therefore the curve for  $K_2$  should be used.

$$r_1/r_2 = 0.875 \text{ and } \theta = 75^\circ$$

from Fig. 10,  $K_{(2)} = K_2 = 2.67$

$$y = 1.64 \frac{(2.67)(0.015)(25.4 \text{ mm})^2}{20 \text{ mm}} = 2.11 \text{ mm (0.083 in)}$$

### Alternate Solution:

This method may be used as a check or in place of using the curves in Fig. 10.

Deflection equation from Table 1, type 2, shape D:  $y = 0.55 \frac{\epsilon l^2}{c_{(3)}}$

The value for  $c_{(3)}$ , which is the distance from the neutral axis to the outermost fiber, can be calculated from the equations shown below:

$$c_2 = r_2 \left[ 1 - \frac{2 \sin \phi (1 - h/r_2 + \frac{1}{2 - h/r_2})}{3 \phi} \right]$$

$$c_1 = r_2 \left[ \frac{2 \sin \phi}{3 \phi (2 - h/r_2)} + (1 - \frac{h}{r_2}) \frac{2 \sin \phi - 3 \phi \cos \phi}{3 \phi} \right]$$

Use  $c_2$  for  $c_{(3)}$ , if the tensile stress occurs in the convex side of the beam. Use  $c_1$  for  $c_{(3)}$ , if the tensile stress occurs in the concave side. For this particular problem, it is necessary to calculate  $c_2$ .

$$c_{(3)} = c_2 = 20 \text{ mm} \left[ 1 - \frac{2 \sin 37.5^\circ}{3 (0.654)} \left( 1 - \frac{2.5 \text{ mm}}{20 \text{ mm}} + \frac{1}{2 - 2.5 \text{ mm}/20 \text{ mm}} \right) \right] = 2.52 \text{ mm}$$

Solving for  $y$  using  $c_2$  yields;

$$y = 0.55 \frac{(0.015)(25.4 \text{ mm})^2}{(2.52 \text{ mm})} = 2.11 \text{ mm (0.083 in)}$$

Both methods result in a similar value for allowable deflection.

## Deflection

In the case of torsion snap joints, the deflection is not the result of a flexural load as with cantilever snaps but is due to a torsional deformation of the fulcrum. The torsion bar (Fig. 21) is subject to shear.

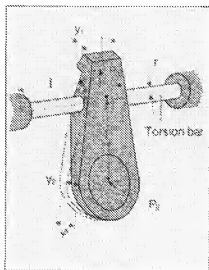


Fig. 21: Snap-fitting arm with torsion bar

The following relationship exists between the total angle of twist  $\varphi$  and the deflections  $y_1$  or  $y_2$  (Fig. 21):

$$\sin \varphi = \frac{y_1 = y_2}{l_1 \quad l_2}$$

where

$\varphi$  = angle of twist

$y_1, y_2$  = deflections

$l_1, l_2$  = lengths of lever arm

The maximum permissible angle  $\varphi_{pm}$  is limited by the permissible shear strain  $\gamma_{pm}$ :

$$\varphi_{pm} = \frac{180}{\pi} \cdot \frac{\gamma_{pm} \cdot l}{r}$$

where

$\varphi_{pm}$  = permissible total angle of twist in degrees

$\gamma_{pm}$  = permissible shear strain

$l$  = length of torsion bar

$r$  = radius of torsion bar

(valid for circular cross section)

The maximum permissible shear strain  $\gamma_{pm}$  for plastics is approximately equal to:

where

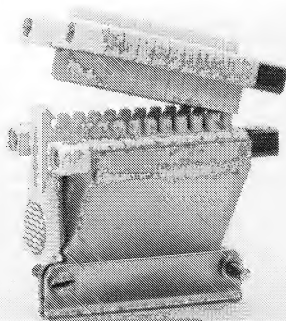
$\gamma_{pm}$  = permissible shear strain

$\epsilon_{pm}$  = permissible strain

$\nu$  = Poisson's ratio (for plastics approx. 0.35)

$$\gamma_{pm} \approx (1 + \nu) \epsilon_{pm}$$

$$\gamma_{pm} \approx 1.35 \epsilon_{pm}$$



## Deflection force

A force  $P$  is required to deflect the lever arm by the amount  $y_{(1,2)}$ . The deflection force can act at points 1 or 2. For example see Fig. 21. In this case,

$$P_1 l_1 = P_2 l_2 = \frac{\gamma G I_p}{r} (x_2)^*$$

where

$G$  = shearing modulus of elasticity

$\gamma$  = shear strain

$I_p$  = polar moment of inertia

$$= \frac{\pi r^4}{2}; \text{ for a solid circular cross section}$$

\*Note: The factor 2 only applies where there are two torsion bars, as in Fig. 21.

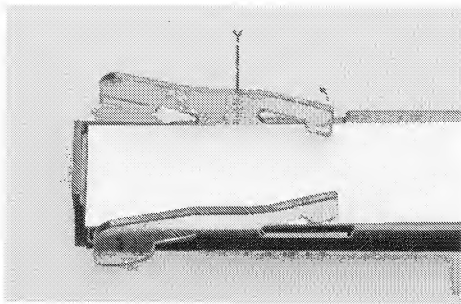
The shear modulus  $G$  can be determined fairly accurately from the secant modulus as follows:

$$G = \frac{E_s}{2(1+\nu)}$$

where

$E_s$  = secant modulus

$\nu$  = Poisson's ratio



*Example of snap-fitting rocker arm (flexure and torsion about the Y axis)*

## Permissible undercut

The annular snap joint is a convenient form of joint between two rotationally symmetric parts. Here, too, a largely stress-free, positive joint is normally obtained. The joint can be either detachable (Figs. 22a, 23), difficult to disassemble or inseparable (Fig. 22b) depending on the dimension of the bead and the return angle. Inseparable designs should be avoided in view of the complex tooling required (split cavity mold).

The allowable deformation should not be exceeded either during the ejection of the part from the mold or during the joining operation.

The permissible undercut as shown in Fig. 24 is limited by the maximum permissible strain

$$Y_{\text{per}} = \epsilon_{\text{pm}} \cdot d$$

Note:  $\epsilon_{\text{pm}}$  is absolute value.

This is based on the assumption that one of the mating parts remains rigid. If this is not the case, then the actual load on the material is correspondingly smaller. (With components of equal flexibility, the strain is halved, i.e., the undercut can be twice as large.)

$W$  = mating force

$y$  = undercut

$\alpha$  = lead angle

$\alpha'$  = return angle

$t$  = wall thickness

$d$  = diameter at the joint

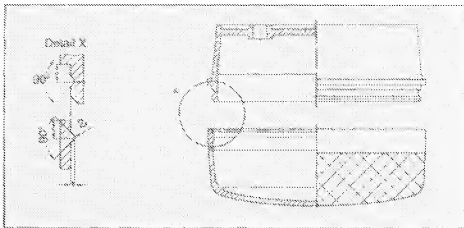


Fig. 23: Annular snap joint on a lamp housing

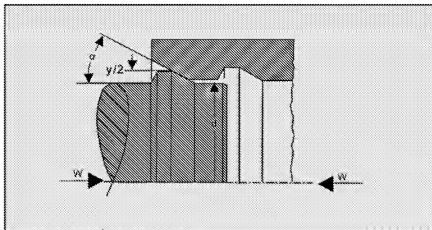


Fig. 24: Annular snap joint—symbols used

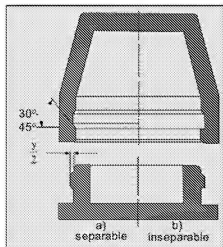
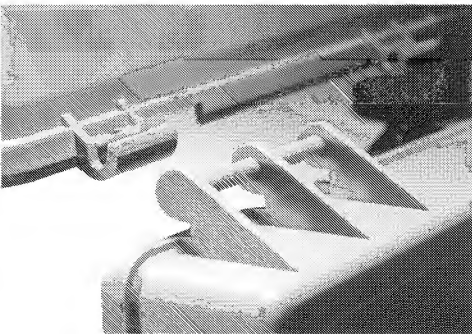


Fig. 22: Annular snap joint



## Deflection force, mating force

The determination of the mating force  $W$  is somewhat more complicated for annular snap joints. This is because the snap-fitting bead on the shaft expands a relatively large portion of the tube (Fig. 25). Accordingly, the stress is also distributed over a large area of the material surrounding the bead.

Experimentally proven answers to this problem are based on the "theory of a beam of infinite length resting on a resilient foundation." Two extreme cases are depicted in Fig. 26.

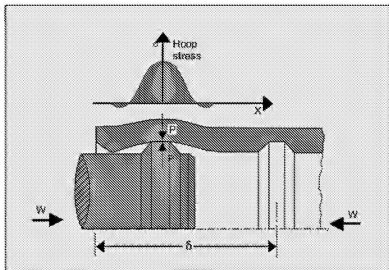


Fig. 25: Stress distribution during joining operation

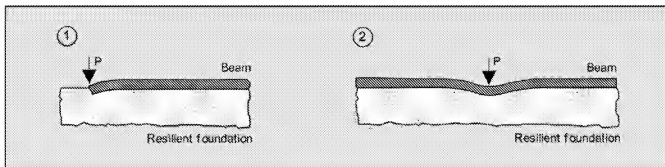


Fig. 26: Beam resting on a resilient foundation

- ① The force  $P$  is applied at the end of the beam. (This corresponds to a snap joint with the groove at the end of the tube.)
- ② The force  $P$  is applied a long distance ( $co$ ) from the end of the beam. (This is equivalent to an annular snap joint with the groove remote from the end of the tube)

A somewhat simplified version of the theory may be expressed as follows for joints near the end of the tube:

$$P = y \cdot d \cdot E_s \cdot X$$

where

$P$  = transverse force

$y$  = undercut

$d$  = diameter at the joint  $E_s$  = secant modulus

$E_s$  = secant modulus

$X$  = geometric factor

The geometric factor  $X$  takes into account the geometric rigidity.

As far as the mating force is concerned, friction conditions and joint angles must also be taken into consideration.

$$W = P \frac{\mu + \tan \alpha}{1 - \mu \tan \alpha}$$

where

$\mu$  = friction coefficient

$\alpha$  = lead angle

The geometric factor, assuming that the shaft is rigid and the outer tube (hub) is elastic, is as follows:

$$X_N = 0.62 \frac{\sqrt{(d_0/d - 1) / (d_0/d + 1)}}{[(d_0/d)^2 + 1] / [(d_0/d)^2 - 1] + \nu}$$

where

$d_0$  = external diameter of the tube

$d$  = diameter at the joint

$\nu$  = Poisson's ratio

If the tube is rigid and the hollow shaft elastic, then

$$X_w = 0.62 \frac{\sqrt{(d/d_i - 1)(d/d_i + 1)}}{[(d/d_i)^2 + 1][d/(d/d_i)^2 - 1] - \nu}$$

where

$d$  = diameter at the joint

$d_i$  = internal diameter of the hollow shaft

The geometric factors  $X_N$  and  $X_w$  can be found in Fig. 27.

The snap joint is considered "remote" if the distance from the end of the tube is at least

$$\delta = 1.8 \sqrt{d} \cdot t$$

where

$d$  = joint diameter

$t$  = wall thickness

In this case, the transverse force  $P$  and mating force  $W$  are theoretically four times as great as when the joint is near the end of the tube. However, tests have shown that the actual mating forces rarely exceed the factor 3

$$P_{\text{remote}} = 3P_{\text{near}}$$

$$W_{\text{remote}} = 3W_{\text{near}}$$

This means that if the joint lies between  $O$  and  $\delta$  minimum, then the factor is between 1 and 3.

The secant modulus  $E_s$  must be determined as a function of the strain  $\epsilon$  from Fig. 16. For the sake of simplicity, it may be assumed here that the strain

$$\epsilon = \frac{y}{d} \cdot 100\%$$

where

$y$  = undercut

$d$  = diameter,

over the entire wall thickness. (In fact, it varies at different points and in different directions on the wall cross section).

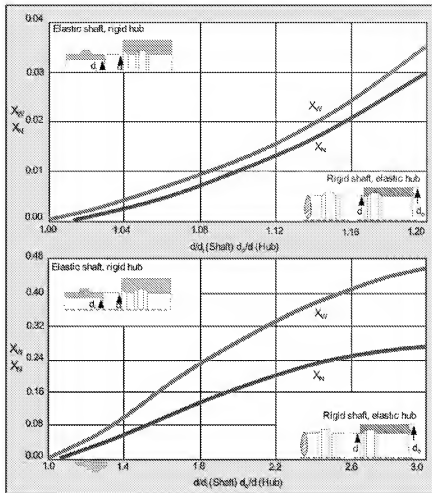


Fig. 27: Diagrams for determining the geometric factor  $X$  for annular snap joints

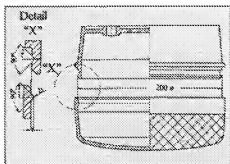


Fig. 28: Lamp housing with cover

## Calculation example annular snap joint

### Given:

Lamp cover and housing made of Makrolon® polycarbonate.  
Snap-fitting groove near the end.

### Dimensions:

$d = 200 \text{ mm}$  (7.87 in)

$t = 2.5 \text{ mm}$  (0.098 in) for both mating parts

$y = 2 \times 1 \text{ mm}$  (0.039 in)

$\alpha = 45^\circ$

The lead and return angle is  $45^\circ$ . The edge is rounded, however, and the effective angle may be assumed to be  $\alpha = 30^\circ$ .

### Required:

Occurring strain  $\epsilon$

Deflection force  $P$

Mating force  $W$

### Solution

#### Strain:

$$\epsilon = \frac{y}{d} \quad \epsilon = \frac{1 \text{ mm}}{200 \text{ mm}} \cdot 100\%$$

$$\epsilon = 0.5\%$$

\*Since both mating parts have approximately equal stiffness, the deflection for each part is approximately half the undercut.

This strain is permissible for polycarbonate according to Table 2.

#### Transverse force $P$ :

$$P = y \cdot d \cdot E_s \cdot X$$

As the mating parts are of approximately equal stiffness, the calculation may be performed for either component. In this case the lamp cover (hub) has been chosen.

$X_N$  from Fig. 27 with

$$\frac{d_0}{d} = \frac{200 + 2 \times 2.5}{200} = 1.025$$

$$X_N = 0.0017 = 1.7 \times 10^{-3}$$

Secant modulus  $E_S$  from Fig. 16

$$E_s = 2,200 \text{ MPa} (320,000 \text{ psi})$$

$$P = 1 \text{ mm} \times 200 \text{ mm} \times 2,200 \text{ MPa} \times$$

$$1.7 \times 10^{-3} = 748 \text{ N}$$

$$P = 748 \text{ N} (168 \text{ lb})$$

Mating force,  $\mu = 0.6$  from Table 3:

$$W = P \cdot \frac{\mu + \tan \alpha}{1 - \mu \tan \alpha} = 748 \text{ N} \times 1.8 = 1,346 \text{ N}$$

Value 1.8 (from Fig. 18)

$$W = 1,346 \text{ N} (302.5 \text{ lb})$$

The mating and opening force  $W$  in this case is a considerable force. It should be remembered, however, that such a force would only occur if mating parts in true axial alignment were joined by machine. In manual assembly, the greater part of the bead is first introduced into the groove at an angle and only the remaining portion is pressed or knocked into position. The mating forces occurring under these circumstances are much smaller, as only part of the bead is deformed.

With all the examples of snap joints mentioned so far, the stiffer of the two mating parts was assumed to be absolutely rigid. Consequently, the more flexible of the two components was theoretically deformed by the full amount of the undercut.

Where both parts are deformable, however, the sum of these deformations is equal to the undercut, i.e., each deformation is smaller.

The mating force and the deformations occurring in two flexible mating parts can be determined most simply by using a graph.

For this purpose, the transverse force for each component is determined as a function of deflection on the assumption that the other component is absolutely rigid; a "deflective curve" is then plotted for each mating part as shown in Fig. 29a and b.

These "deflection curves" are then superimposed (Fig. 29c). The point of intersection of the two curves gives the actual deflection force  $P$  and the deflections  $y_1$  and  $y_2$ .

With the aid of these quantities  $P$ ,  $y_1$  and  $y_2$ , the individual strains and the mating force can then be determined without difficulty, as described earlier.

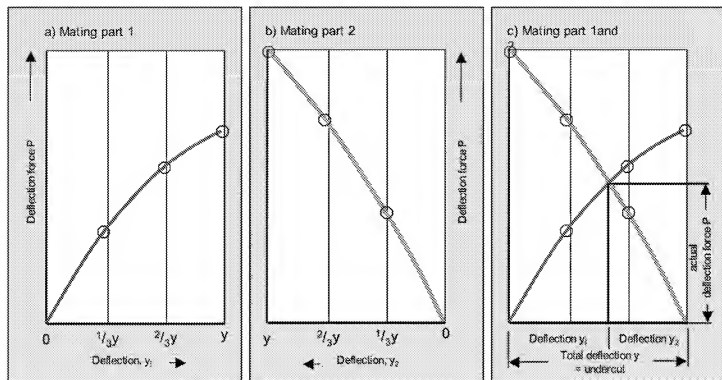
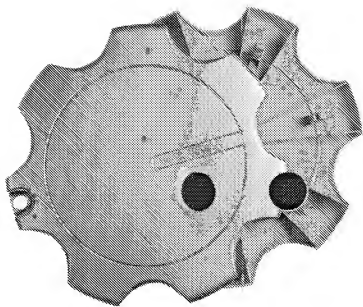


Fig. 29: Determination of deformation and transverse force when both mating parts are flexible

a	dimensions	$\alpha$	angle of inclination
b	dimensions	$\alpha'$	return angle
c	distance between outer fibre and neutral fibre	$\delta$	distance of snap-fitting groove from the end
d	diameter at the joint	$\epsilon$	strain
$d_i$	internal diameter	$\epsilon_{pm}$	maximum allowable strain
$d_o$	external diameter	$\epsilon_{alt}$	strain at break
$E_o$	modulus of elasticity (intrinsic tangential modulus)	$\epsilon_y$	yield strain
$E_s$	secant modulus	$\varphi$	angle of twist
F	friction force	$\gamma$	shear strain
G	shear modulus	$\mu$	friction coefficient
H	height thickness at the root	$\nu$	Poisson's ratio
$I_p$	polar moment of inertia	P	angle of repose
K	geometric factor for ring segments	$\sigma$	stress
l	length, length of lever arm	$\theta$	arc angle of segment
N	normal force due to insertion		
P	deflection force		
R	resultant insertion force		
r	radius		
t	wall thickness		
W	mating force		
X	geometric factor for annular snap joint		
	index W=shaft		
	index N=hub		
y	undercut, deflection		
Z	axial section modulus		

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